HEAT TRANSFER AND PRESSURE DROP FOR AIR FLOW THROUGH AN ENHANCED PIPE

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SHASHI KANT KHANNA

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CERTIFICATE

This is to certify that the thesis work entitled "HEAT TRANSFER AND PRESSURE DROP FOR AIR FLOW THROUGH AN ENHANCED PIPE" has been carried out by Shashi Kant Khanna under my supervision and has not been submitted elsewhere for the award of a degree.

Keshav Kant

May 10, 1993

(Dr. KESHAV KANT)

Associate Professor

Department of Mechanical Engineering

Indian Institute of Technology

Kanpur 208016

INDIA

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CONTENTS

ABSTRACT

The present experimental study was taken up to investigate the augmentation in heat transfer and the increase in pressure drop by the use of in-line propellers in single phase flow. The data were taken in the range of 2300<Re<20000. The in-line propellers and their mounts were housed in plexiglass pipe sections for flow visualization. Each plexiglass section was of 50 mm length X 25.4 mm i.d. and on either side was connected to brass pipe sections of the same diameter, but of 300 mm in length. Brass pipe sections were subjected to constant heat flux. Temperatures of air at various locations inside the test section and the wall temperatures were measured using copper-constantan thermocouples. Pressure drops were measured using an inclined tube manometer. Flow velocity was measured with the help of a Pitot tube. Measurements were made with and without propellers. Both the heat transfer coefficient and the associated pressure drops were compared with those for an unenhanced tube. Fractional increase in average Nusselt number, fractional increase in Fanning friction factor and efficiency were determined as a functions of Reynolds number and number of propellers. For the range of Re used, the efficiency for the 3 propellers placed in the flow was found to be lower than that for twisted tapes and several spirally shaped geometries of fluted, ribbed/finned and indented types. But it was found to be higher than that with mesh inserts, brush inserts, disks, streamed line shapes, static mixers and propeller type baffles.

NOMENCLATURE

A_c cross sectional area of flow, m^2

A_s inner surface area of the heated pipe, m^2

C_h constant depending upon the geometry of the enhanced passage

c_p specific heat, J kg^{-1} K^{-1}

D inside diameter of the pipe, m

D_o outside diameter of the pipe, m

f Fanning friction factor

f_e augmented Fanning friction factor in presence of propellers

f_{fi} fractional increase in the friction factor (f_e/f_s)

f_s Fanning friction factor for the smooth tube

g acceleration due to gravity, m s^{-2}

h heat transfer coefficient, W m^{-2} K^{-1}

k thermal conductivity, W m^{-1} K^{-1}

L length of the heated section, m

L_t distance between the two manometric taps, m

$\dot{m}$ mass flow rate, kg s^{-1}

n number of propellers in the test section

Nu_e average enhanced pipe Nusselt number

Nu_{fi} fractional increase in the average Nusselt number (Nu_e/Nu_s)

Nu_s average Nusselt number for smooth pipe

Nu_x local Nusselt number

q input heat flux, W m^{-2}

Q rate of heat input, W

Re Reynolds number

Re^+ roughness Reynolds number

Re_η Reynolds number corresponding to maximum efficiency

St_e Stanton number for enhanced pipe

St_s Stanton number for smooth pipe

T_{ax} air temperature at location x, °C

T_b average bulk air temperature, °C

T_i inlet bulk temperature of the test fluid, °C

T_o outlet bulk temperature of the test fluid, °C

T_{wx} outside wall temperature as measured at location x, °C

x distance from the point where the heating begins, m

x/D dimensionless axial distance

x_d hydrodynamic entry length, m

Greek Symbols

ρ_a density of air, kg m^{-3}

ΔP pressure drop between the two taps of the manometer, N m^{-2}

η efficiency of the turbulence promoter

ν_b kinematic viscosity at mean bulk temperature, m^2 s^{-1}

Additional Subscripts

a arbitrary condition

c,a critical parameter for arbitrary condition

c,r critical parameter for reference condition

m reduced parameters

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CHAPTER 1

INTRODUCTION

1.1 MOTIVATION

Augmentation of convective heat transfer and improvement of the efficiency of the devices, wherein the heat transfer process occurs, are the principal goals in the design and development of heat exchangers. This is particularly important for gas heat exchangers where the heat transfer rates are typically low.

Enhancement of heat transfer leads to decrease of over all dimensions and weight of the heat exchanger as compared to their magnitudes under ordinary conditions. Augmentation of heat transfer plays a critical role in space and aeronautical applications where weight and size of the heat exchanger are of crucial importance.

In case of heat transfer taking place in a viscous flow over a surface, the main thermal resistance is exerted by the boundary layer, which develops on the surface. The thicker the boundary layer and the lower the thermal conductivity of the flowing fluid, the lower will be the heat transfer coefficient. Therefore, to enhance the convective heat transfer, the thermal resistance in the boundary layer must be reduced. This can be accomplished by basically agitating the flow rather than increasing the heat transfer surface area. The techniques used for agitating the flow are described in the following chapter.

In order to be effective, augmentation technique must be applied to the dominant thermal resistance in the system. It

should preferably be simple to use and should reduce the capital cost and increase the thermal rating of the heat exchanger.

A number of geometrical arrangements when placed in the flow generate the swirl flow. Any device, which can create swirl flow and yet present a smaller area for drag, may be preferred as a turbulence promoter. It may augment the convective heat transfer coefficient, but not substantially increase the pressure drop as compared to a smooth tube carrying no augmenters of any kind in the flow. One such device which has been studied to some extent [Betal (1989); Chaturvedi (1992)] is in-line propeller. In order to establish its superiority or otherwise as a heat transfer augmenter is the subject of the thesis.

1.3 SCOPE OF THE PRESENT WORK

The objectives of the present experimental work are

1. Design of a test rig to determine the heat transfer coefficient and pressure drop for air flow through a pipe with and without the in-line propellers.
2. Comparison of the results of the enhanced tube with that of the smooth tube.
3. Comparison of the performance of in-line propellers with that of other augmenting devices and augmenting flow passages.

In order to visualize the propeller rotation, each propeller is mounted in a plexiglass section. The fact that plexiglass can not withstand high temperature, limited the heat flux input in the experiments to approximately to 2700 W/m^2.

CHAPTER 2

GENERAL BACKGROUND

2.1 INTRODUCTION

The goal of any heat exchanger design is to do more with less whether that is to design a more compact heat exchanger or to increase the heat duty for a given size of heat exchanger or to achieve some other specific objective subjected to some specific constraints.

The convective heat transfer coefficient in a heat exchanger can be augmented by creating interruptions in the flow passages. The increase in the heat transfer coefficient is accompanied by an increase in the pressure drop or pumping power. One obvious advantage is the reduction in the amount of heat transfer surface material and therefore a lower capital cost and a relatively more compact heat exchanger. The area of heat transfer which is related to increasing the convective heat transfer coefficient either by placing obstruction in the path of flow or designing the heat transfer surface in a special manner refers to enhanced or augmented heat transfer.

There have been many literature surveys and reviews on augmentation of convective heat transfer [Bergles (1972), Bergles (1976), Nakayama (1982), Webb (1981) and Lazarek (1980) Reay (1991), Obot et al. (1992) and Webb (1987)]. These surveys have generally discussed the technical aspects of enhanced heat transfer coefficient and friction factor or pressure drop in the

enhanced tube relative to that for smooth tube under the same conditions. Some papers have discussed methods to evaluate the performance of augmented heat exchanger when geometric and operating constraints are imposed.

Bergles et al. (1983) compiled a list of over 3000 papers and reports, and webb et al. (1983) compiled a list of over 450 U.S. patents in this area. Bergles et al. (1984) also compiled a list of over 200 companies which are associated with enhanced tubing, enhanced heat exchangers or enhancement related products.

2.2 AUGMENTATION TECHNIQUES

The Enhancement in the convective heat transfer can be achieved by using active, passive or compound techniques.

1. **PASSIVE TECHNIQUES**

No external energy is required to effect the augmentation by these techniques. Some of these are as follows

a. Surface roughness

b. Internal extended surface

c. Displaced promoters

d. Swirl flow devices

e. Additives

f. Treated surfaces

g. Coiled tubes

h. Surface tension devices

2. **ACTIVE TECHNIQUES**

These require an additional source of energy to augment heat transfer coefficient. According to the type of device used these

techniques can be further classified as

a. Mechanical aids

b. Heat surface vibration

c. Fluid pulsation

d. Electrostatic fields

e. Suction and injection

f. Jet impingement

3. COMPOUND TECHNIQUES

These techniques consist of simultaneous application of at least two separate techniques of convection heat transfer enhancement.

a. Rough tube wall with twisted tape inserts

b. Rough cylinder with acoustic vibrations

c. Internally finned tube with twisted tape inserts

d. Finned tube with fluidized bed

e. Externally finned tube subjected to vibration

f. Finned tube with an electric field

g. Fluidized bed with pulsations of air

More than 88% of the patents listed in Webb et al. (1983) involve passive techniques and about 40% of these are for two phase heat transfer. The main reason behind not using active techniques that much is mechanical difficulty involved in their construction.

2.3 PERFORMANCE EVALUATION OF AUGMENTATION TECHNIQUES

The evaluation of augmentation techniques depends upon the effect of enhancement on

a. Heat duty

b. Pressure drop or pumping power

c. Surface area

The main emphasis of the augmentation technique is on the extent of increase in the convective heat transfer coefficient whereas the accompanying pressure drop has to be tolerated.

Webb (1981) provided the following design objectives in his development of Performance Evaluation Criteria (PEC) for single phase flows in fluid to fluid heat exchangers using enhanced surfaces.

1. Reduced amount of heat transfer surface material and therefore lower capital cost and relatively more compact heat exchanger for equal pumping power or heat duty.

2. Increased heat duty or decreased inlet temperature difference for equal pumping power and fixed total length of heat exchanger tubing.

3. Reduced pumping power for equal heat duty and total length of heat exchanger tubing.

Depending upon the application of the enhanced surface in place of a smooth surface, any of the above objectives might be the goal of the heat exchanger design. When additional design constraints like fixed flow area, fixed or variable geometry are imposed then a specific PEC can be developed and evaluated to see if a particular enhancement technique satisfies the design objectives (Jensen, 1988).

one of the methods used to compare the performance of different types of turbulence promoters is to use the efficiency

defined by [Webb et al. 1971; Gee et al. 1980]

$$\eta = (St_e/St_s)/(f_e/f_s) = (Nu_e/Nu_s)/(f_e/f_s) \qquad (2.1)$$

This index is usually plotted against roughness Reynolds number defined by

$$Re^+ = (e/D).Re.(f_e/2)^{0.5} \qquad (2.2)$$

Sometimes this efficiency may be plotted against Re and in many cases it is difficult to draw an accurate conclusion of general validity from the results regarding the performance of the various passages because of the complexities introduced by the transition process [Obot and Esen 1992]. Instead reduced efficiency can be used for performance evaluation which is defined as

$$\eta_m = (Nu_m/Nu_s)/(f_m/f_s) \qquad (2.3)$$

Where,

Reduced Nusselt number

$$Nu_m = (Nu_{c,r}/Nu_{a,r}).Nu_a \qquad (2.4)$$

Reduced friction factor

$$f_m = (f_{c,r}/f_{a,r}).f_a \qquad (2.5)$$

This reduced efficiency can be plotted against Reduced Reynolds number defined as

$$Re_m = (Re_{c,r}/Re_{c,a}).Re_a \qquad (2.6)$$

The use of Reduced parameters does not entirely eliminate the above difficulty, but gives a satisfactory treatment of data.

2.4 THERMAL PERFORMANCE OF VARIOUS PASSIVE DEVICES

A large number of turbulence promoters of various geometrical configuration have been studied by Bergles (1973). Twisted tapes were most extensively studied, both experimentally and

theoretically [Bergles, 1974; Smithberg and landis, 1964; Burfort and Rice, 1983]. Other devices that have been studied are static mixers [Colburn and king, 1931; Lin et al., 1977], wire coils [Krieth and Margolis, 1959; Setumadhavan and Raja Rao, 1983], disks and streamlined shapes [Evans and Churchill, 1962], detached promoters [Thomas, 1967], mesh and brush inserts [Megerlin et al., 1974].

Betal (1989) conducted experiments with three propellers placed in a staggered manner in a pipe with water as a test fluid. The results of the experiments showed that the heat transfer coefficient increased by 10 to 100% at Re>4000 and pressure drop increased from 3.5 to 4.75 times of the corresponding value for the smooth tube. Chaturvedi and Kant (1992) conducted experiments with in-line propellers in the range 16000<Re>68000 using water as a test fluid. Results showed that maximum heat transfer augmentation was obtained at the interpropeller distance of 9.72 D and Re=45000.

Regardless of the technique used to augment the heat transfer process or whether it is applied to single phase or two phase heat transfer, the heat transfer enhancement data are compared with those for smooth tube data obtained at similar operating conditions in order to see the benefits associated with the different augmentation techniques. Table 2.1 compares the heat transfer augmentation, efficiency and the increase in the friction resulting from the use of some passive promoters.

TABLE 2.1

Comparison of Heat Transfer Coefficient Augmentation with Various Augmenting Devices under their Best Operating Conditions.

promoter	Exp. Conditions	Re	Nu_{fi}	f_e/f_s	η
Twisted tapes (Laminar) Honk and Bergles (1974)	For water in 1.02 cm i.d. tube with 0.97 cm wide twisted tapes of pitch to dia ratio of 2.45	100 2000	1.3 8.5	11.5 11.5	0.74 to 0.11
Twisted tapes (Turbulent Region) Smithberg et al. (1964)	For water in 3.51 cm i.d. tube with 3.48 cm wide twisted tapes of pitch to dia ratio of 3.62	20000 50000	3.5 3.5	2.5 2.0	1.5 to 1.4
Wire coils Krieth et al. (1959)	For water in 1.35 cm i.d. tube with close fitting wire coils of pitch to dia ratio of 1.77	10000 10000	3.2 3.4	6.8 4.8	0.71 to 0.47
Mesh inserts Megerlin et al. (1974)	For water in 0.53 cm i.d. tube with close fitting s.s. mesh inserts of 80% porosity.	10000 to 30000	9.0	55.0	0.16
Brush inserts Megerlin et al. (1974)	For water in 0.53 cm i.d. tube with close fitting s.s. brushes	10000 to 30000	5.0	35.0	0.09
Disks Evans and Churchill (1962)	For water in 2.55 cm i.d. tube with disks of 2.33 cm dia at a spacing of 12 dia	10000 50000	2.5 2.0	75.0 100.0	0.04 to 0.02
Promoters Thomas (1967)	cm i.d. tube with 1.91 cm o.d. and 1.57 i.d. rings spaced 0.64 cm apart.	30000	1.9	12.5	0.15 to 0.03

promoter	Exp. Conditions	Re	Nu_{fi} #	f_{fi} *	η
Streamline shapes Evans and Churchill (1962)	For water in 2.55 cm i.d. tube with streamline shapes of 2.23 cm dia at a uniform spacing of 12 tube dia.	10000 5000	2.2 1.7	35.0 40.0	0.06 to 0.04
Static Mixers Lin et al. (1977)	For R113 in 1.27 cm i.d. tube with close fitting static mixers of pitch to dia ratio of 1.5	1000 5000	6.0 10.0	3.0 4.6	2.28 to 2.0
Propeller type baffles (static) colburn et al. (1931)	For air in 6.67 cm i.d. tube with propeller type baffles spaced uniformly 7.62 cm apart.	25000 to 150000	3.4	40.0	0.09
In-line Propellers Betal (1989)	For water in 5.0 cm i. d. tube with 3 in-line propellers placed staggeredly.	4000 to 10000	1.1 to 2.0	3.5 to 4.7	0.5
In-line Propellers Chaturvedi and Kant (1992)	For water in 5.0 cm i. d. tube with 5 in-line propellers spaced 9.72 D apart.	40000 to 55000	6.0	4.0	2.5

$Nu_{fi} = Nu_e/Nu_s$

* $f_{fi} = f_e/f_s$

Obot (1992) tested 23 commercially available enhanced passages for heat transfer augmentation and pressure drop over the entire range of flow conditions extending from Re=400 to Re=50000. These geometries are summarized in Table 2.2 and are shown in Fig. 2.1. These enhanced passages were spirally shaped and were of fluted, ribbed/finned and indented types. Test conditions for

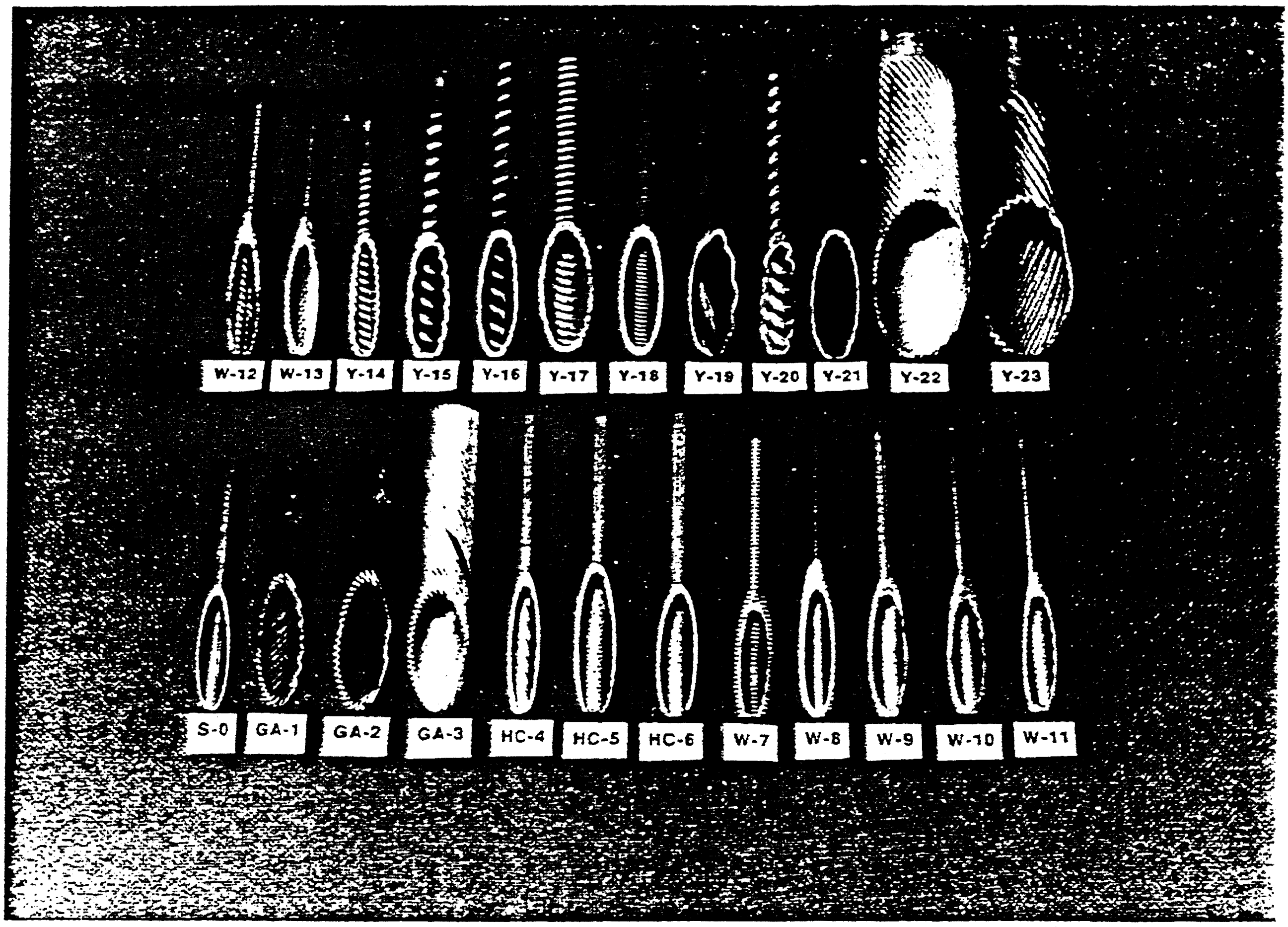


FIGURE 2.1 Photograph of Spiral Flow Passages

Tube	D (mm)	t (mm)	e (mm)	N_s	l (mm)	p (mm)	α (degrees)	e/D	p/e	Material	Roughness Description
S-0	13.39	1.2								Copper	Smooth
GA-1	21.45	0.7	0.95	20	82.0	4.1	39.4	0.044	4.3	Stainless Steel	Spirally fluted
GA-2	23.96	0.98	1.33	25	141	5.6	28.1	0.056	4.2	Stainless Steel	Spirally fluted
GA-3	28.49	1.89	1.58	31	160	5.2	29.2	0.055	3.3	Aluminum	Spirally fluted
HC-4	13.87	1.0	0.3	10	82.0	8.2	28.0	0.022	27.3	Copper	Spirally ribbed
HC-5	17.78	0.64	0.5 (0.3)	25 (25)	142.5 (93.0)	5.7 (3.7)	21.5 (-31.0)	0.028 (0.017)	11.4 (12.3)	Copper	3-D Spirally ribbed
HC-6	17.61	0.72	0.26 (0.14)	25 (25)	140.0 (92.1)	5.6 (3.7)	21.5 (-31)	0.015 (0.008)	21.5 (26.4)	Copper	3-D Spirally ribbed
W-7	14.10	1.07	0.42	1	2.2	2.2	87.2	0.030	5.2	Copper	Spirally ribbed
W-8	14.40	1.12	0.10	1	1.0	1.0	88.7	0.007	10.0	Copper	Spirally ribbed
W-9	15.90	1.52	0.5	41	102.5	2.5	26.0	0.032	5.0	Copper	Spirally ribbed
W-10	14.95	1.48	0.55	25	100.0	4.0	25.2	0.037	7.3	Copper	Spirally ribbed
W-11	14.45	1.49	0.45	20	76.0	3.8	30.9	0.031	8.5	Copper	Spirally ribbed
W-12	14.56	1.59	0.50	10	40.0	4.0	48.8	0.034	8.0	Copper	Spirally ribbed
W-13	14.45	1.50	0.51	25	120.0	4.8	20.7	0.035	9.4	Copper	Spirally ribbed
Y-14	12.68	1.08	0.38	3	15.0	5.0	69.4	0.030	13.2	Copper	Spirally indented
Y-15	19.16	1.24	1.27	3	30.0	10.0	63.5	0.066	7.9	Copper	Spirally indented
Y-16	19.53	1.23	0.51	3	30.0	10.0	63.9	0.026	19.6	Copper	Spirally indented
Y-17	24.22	1.67	0.31	3	15.0	5.0	78.8	0.013	16.1	Copper	Spirally indented
Y-18	18.81	1.66	0.36	3	7.6	2.6	82.7	0.019	7.2	Copper	Spirally indented
Y-19	22.88	1.04	1.5	6	19.8	3.3	74.6	0.066	2.2	K10	Spirally indented
Y-20	16.05	1.36	1.0	3	30.0	10.0	59.2	0.062	10.0	Copper	Spirally indented
Y-21	23.45	0.93	0.52	1(6)	6.0	6.0	85.3	0.022	11.5	K10	Spirally indented Doubly enhanced
Y-22	48.65	0.87	2.0	43	273.1	6.35	29.2	0.041	3.2	YAB	Spirally fluted
Y-23	47.67	1.29	2.96	25	277.5	11.1	28.4	0.062	3.8	Copper	Spirally fluted

TABLE 2.2 Geometric Characteristics of Enhanced Flow Passages

Tube	D (mm)	L_e (mm)	$L_{e,r}$ (mm)	L (mm)	L_t (mm)	A_s (mm^2)	A_c (mm^2)	L_e/D	$L_{e,r}/D$	L/D	L_t/D
S-0	13.39	2019			381		140.8	150.8			
		1969		458.0	381	19266		147.1		34.2	28.5
					1565.3		361.4				73.0
GA-1	21.45	254			1354.1			11.8			63.1
		1016			920.8			47.4			42.9
GA-2	23.96				1587.5		450.9				
		254			1343			10.6			56.1
		254			2773.4			8.9			97.3
GA-3	28.49	1089			920.75			38.2			32.3
		1089	38.1		838.2			38.2	1.3		29.4
			1524	381.0	304.8	34101	637.5		53.5	13.4	10.7
HC-4	13.87		1968.5	441.1	365.0	19221	151.1		141.9	31.8	26.3
HC-5	17.78		1968.5	344.2	268.0	19224	248.3		110.7	19.4	15.1
HC-6	17.61		1968.5	347.5	271.3	19224	243.6		111.8	19.7	15.4
W-7	14.10		1165.2	434.1	357.9	19229	156.2		82.7	30.8	25.4
W-8	14.40		1968.5	424.9	348.7	19224	162.9		136.7	29.5	24.2
W-9	15.90		1644.7	384.8	308.6	19222	198.6		103.4	24.2	19.4
W-10	14.95		2087.6	410.5	333.5	19278	175.5		139.6	27.4	22.3
W-11	14.45		2089.2	423.4	347.2	19222	164.0		144.6	29.3	24.0
W-12	14.56		2133.6	420.4	344.2	19228	166.5		146.6	28.9	23.7
W-13	14.45		2133.6	423.4	347.2	19222	164.0		147.6	29.3	24.0
Y-14	12.68		1651	482.9	406.7	19227	126.2		130.3	38.9	32.1
Y-15	19.16		1651	319.6	243.4	19231	288.2		86.2	16.7	12.7
Y-16	19.53		1651	313.3	237.1	19223	299.6		84.5	16.0	12.1
Y-17	24.22		1651	448.6	372.4	34131	460.7		68.2	18.5	15.4
Y-18	18.81		1651	325.3	249.1	19224	277.9		87.8	17.3	13.2
Y-19	22.88		1651	474.5	398.3	34110	411.2		72.2	20.7	17.4
Y-20	16.05		1651	381.2	330.4	19223	202.3		102.9	23.8	20.6
Y-21	23.45		838.2	463.1	386.9	34113	431.7		35.8	19.8	16.5
Y-22	48.65		522.3	457.2	381.0	69878	1859		10.7	9.4	7.8
Y-23	47.67		1622.4	466.5	390.2	69855	1785		34.0	9.8	8.2

TABLE 2.3 Test Conditions for Enhanced Flow Passages

TABLE 2.4

Summary of Results for Enhanced Flow Passages

Reynolds number range = 400 - 50000

Flow Passage	f_{fi}	Nu_{fi}	η	Re_η
GA -1	1.3 - 3.0	-	-	-
GA -2	1.3 - 3.1	-	-	-
GA -3	1.3 - 2.6	1.5 - 2.5	0.3 - 1.3	900
HC -4	1.0 - 1.5	0.9 - 1.2	0.5 - 1.0	1000
HC -5	1.0 - 3.0	1.1 - 2.3	0.2 - 1.3	700
HC -6	1.0 - 1.5	0.9 - 1.5	0.3 - 1.2	700
W -7	0.7 - 3.1	0.6 - 2.0	0.1 - 1.0	1500
W -8	0.9 - 1.5	0.6 - 1.5	0.8 - 1.0	1100
W -9	0.8 - 2.8	0.7 - 2.0	0.6 - 1.0	1500
W -10	0.6 - 2.5	0.6 - 2.0	0.2 - 1.5	1500
W -11	0.6 - 2.0	0.7 - 1.9	0.4 - 1.0	1700
W -12	1.1 - 2.5	0.8 - 1.7	0.1 - 0.6	700
W -13	1.0 - 2.2	0.8 - 2.0	0.4 - 1.0	15000
Y -14	1.5 - 4.0	1.0 - 2.5	0.1 - 0.6	800
Y -15	2.0 - 4.0	1.5 - 3.0	0.1 - 0.8	600
Y -16	1.2 - 2.5	1.1 - 2.0	0.8 - 1.5	500
Y -17	1.2 - 3.0	1.1 - 2.5	0.5 - 1.4	900
Y -18	1.0 - 2.0	1.0 - 1.6	0.8 - 1.4	1000
Y -19	1.0 - 2.0	1.5 - 2.0	0.6 - 1.3	13000
Y -20	1.7 - 4.0	1.0 - 2.7	0.1 - 0.6	500
Y -21	1.6 - 2.5	1.5 - 2.5	0.6 - 1.1	600
Y -22	1.5 - 3.0	1.2 - 2.5	0.2 - 1.3	600
Y -23	1.6 - 3.0	1.5 - 2.5	0.4 - 1.0	12000

$Nu_{fi} = Nu_e/Nu_s$

$f_{fi} = f_e/f_s$

Re_η = Reynolds number at which the efficiency is maximum

the flow passages are given in Table 2.3. The results are summarized in Table 2.4.

2.5 SOME RECENT STUDIES ON PASSIVE DEVICES

Liou et al. (1993) performed a numerical and experimental analysis to investigate the heat transfer and fluid flow behavior in a rectangular channel flow with stream wise periodic ribs mounted on one of the principal wall.

Vulchanov et al. (1989) proposed a model to compute the fanning friction factor and the heat transfer coefficient for hydrodynamically and thermally stabilized single phase incompressible turbulent flow in a round tube with internally sand-roughened walls for moderate Prandtl numbers. Both constant heat flux and constant wall temperature boundary conditions are stimulated. A good agreement between the predicted and experimental values is reported.

For the turbulent flow through internally finned tubes important contribution to the experimental data were made by Hilding and Coogan (1964), Lipet et. al. (1969), Bergles et. al. (1971), Watkinson et. al. (1975) and carnovas et. al. (1980). Mikic (1992) investigated turbulent heat transfer augmentation using micro scale disturbances inside a viscous sub layer. The results indicate that for optimal placement of micro disturbances a matching of geometric perturbation with the sub layer scale is required.

Models for predicting the various quantities of interest in the pipe flow have also been proposed. Khalatov (1977) had

proposed a law for heat transfer with swirl flow in the entrance section. A predictive model for the decay of swirl, downstream of the swirl inducer has been developed by Algifri et al. (1987). Algifri et al. (1988) measured heat transfer coefficient along a heated pipe for decaying swirl flow generated by radial blade cascade. These values when compared with the heat transfer coefficient computed from an expression proposed by them for the same showed good agreement.

2.6 ENHANCEMENT STUDIES ON TWO PHASE HEAT TRANSFER

The first survey of enhanced two phase heat transfer covering boiling and condensation was made by Bergles (1976). Lazarek (1980) reviewed forced convective boiling and condensing in horizontal tubes while webb (1981) reviewed enhanced nucleate boiling with special surface geometries.

Bell et al. (1968) studied forced convection boiling of water at atmospheric pressure. Chukhman et al. (1972) used Helium, R-12, R-22 as the working fluid with various coil diameters. They found that the boiling heat transfer coefficient increased with increasing mass velocity, heat flux and ratio of tube diameter to coil diameter. Fractional increase in pressure drop was greater than the fractional increase in the heat transfer coefficient.

For in-tube forced convection boiling, twisted tape inserts are the most extensively studied augmentation technique. Foure et al. (1965) obtained improvement of 25 to 40% in heat transfer for the same pumping power. Jensen and Bensler (1986) tested three twisted tapes in a uniformly heated test section and determined

that swirl flow heat transfer coefficients were up to 90% larger than the axial flow heat transfer coefficients at the same flow conditions. Jensen (1986) developed many two phase boiling heat transfer and pressure drop correlations.

Lavin et al. (1965) studied the effect of integral, internal fins on forced convection boiling. The average heat transfer coefficients with the four finned tubes were 50 to 75% higher than those for smooth tubes. The increase was attributed to the sharp corners associated with each fin providing favorable condition for nucleation. Schlünder et al. (1967) tested star inserts with evaporating R-11. They found that heat transfer coefficient increased slightly as compared to the bare tube. Kubanek et al. studied evaporation of R-22 in integral finned tubes at about 5 atmospheres. The average heat transfer coefficients for the finned tube showed improvement of 600% compared to a smooth tube.

In a recent study, three augmentation techniques namely high finned tubes, micro fin tubes and twisted tape inserts were investigated experimentally for evaporation of R-113 by Reid et al. (1991). The results showed that for internally finned tubes the fractional increase in the Nusselt number varied from 1.1 to 2.8. It was also found that internally finned tube having helix spiral angle of 16 Deg. produced largest enhancement and tube with twisted tape inserts had an enhancement factor of 1.5.

Megerlin et al. (1974) tested mesh and brush inserts in subcooled boiling. The heat transfer coefficient with the inserts was about twice that of the smooth tubes. Leontev et al. (1983) attached a wire gauge to the inner surface of the tube. The heat

transfer coefficient was 1.2 to 2.7 times that of unenhanced tube. Czikk et al. (1981) used a metallic porous coating on the inner surface of the tube. The heat transfer coefficients were 8 to 10 times as compared to the smooth tube.

Royal et al. (1978) studied intube condensation of steam with twisted tape swirl generator. Heat transfer coefficient increased upto 30% compared to smooth tube. Pressure drop increased upto 2.5 times that of smooth tubes. Reisbig (1974) condensed R-12 in several finned tubes and obtained increase in heat transfer coefficient of upto about 300% compared to a smooth tube. Luu et al. (1979) studied condensation of steam and R-113 in internally finned tubes and obtained 120 to 150% increase in heat transfer coefficient.

Yang et al. (1991) conducted experiments to determine the nucleate pool boiling heat transfer performance of a copper-graphite composite surface in R-113. It was found that the boiling heat transfer coefficients on the composite surface were higher than those on the copper surface by a factor of over 6 to 3, respectively at lower and higher values of superheating.

CHAPTER 3

TEST RIG AND PROCEDURE

3.1 INTRODUCTION

A test rig has been designed and constructed for studying the heat transfer augmentation with in-line propellers placed in a pipe flow. The details of the test rig and the experiments are described in the following sections.

3.2 MAIN COMPONENTS AND INSTRUMENTATION

Fig. 3.1 shows the schematic diagram of the test rig which consists of test section and the entry length of 50 D for flow development. The manometer as shown in the Fig. 3.1 is used to measure the pressure drop. The test section is subjected to a constant heat flux. The temperatures of the test section wall and air are measured using Copper-Constantan thermocouples. In the present experiment blower is used to cause the air flow through the pipe.

ENTRY LENGTH

A 1.4 m long X 0.0254 m i.d. brass pipe before the test section is provided for the entry length.

TEST SECTION

It consists of brass pipe sections of 300 mm length and 25.4 mm i.d. and plexiglass sections of 50 mm length and 25.4 mm i.d fixed with propellers as shown in Fig. 3.2. Flanges are used to connect the brass and plexiglass sections. Heating tape of 25 mm

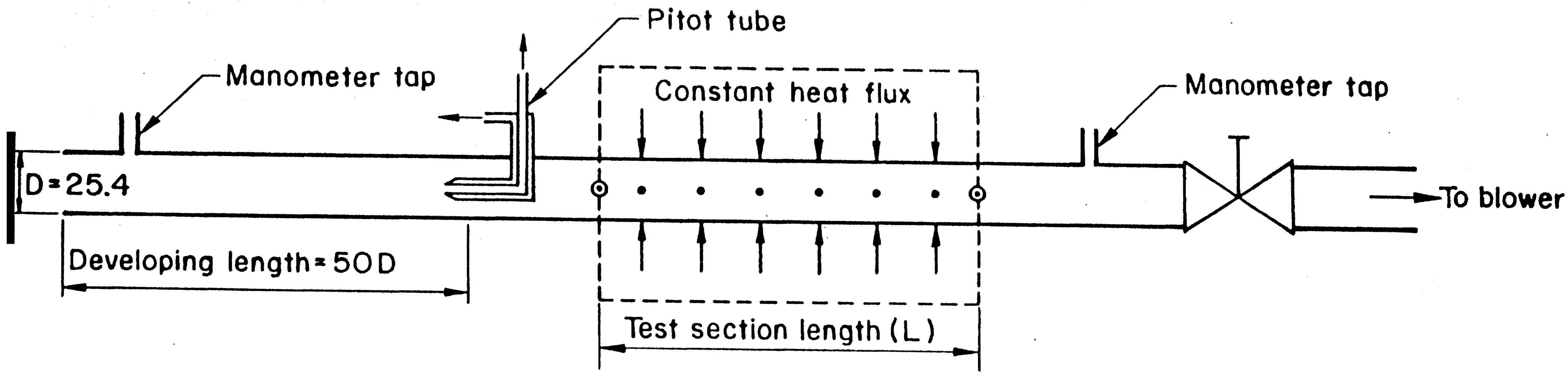
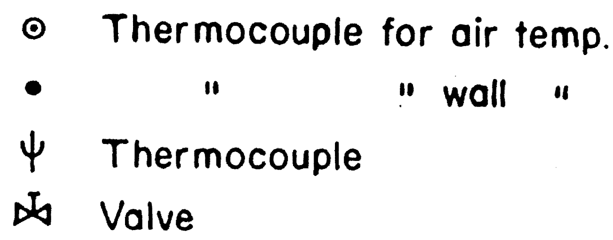


FIGURE 3.1 Schematic Diagram of the Test Rig



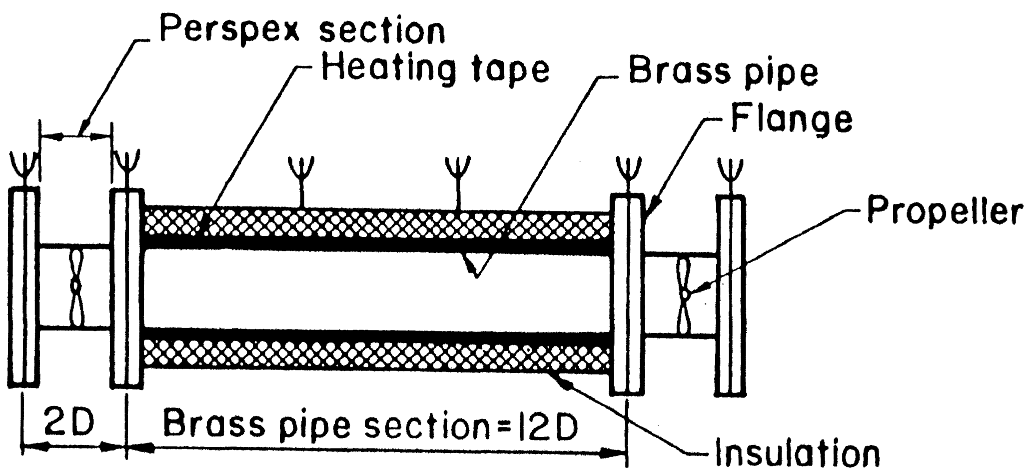


FIGURE 3.2 Schematic Diagram of the Test Section

width is uniformly wound on the brass section to obtain a constant heat flux along the length of the test section. The test section is insulated to minimize the heat loss. Heating of the test section is controlled by varying the voltage across the heating tape using a variac.

IN LINE PROPELLERS

These are fixed in a plexiglass sections of 50 mm length and 25.4 mm i.d., using brass mounting. These brass mountings are shown in Fig. 3.3. It consists of end supporting legs and screws for fixing the legs inside the pipe. The propeller assembly is shown in Fig. 3.4. Complete apparatus is shown in Fig. 3.5.

PROPELLER BLADE

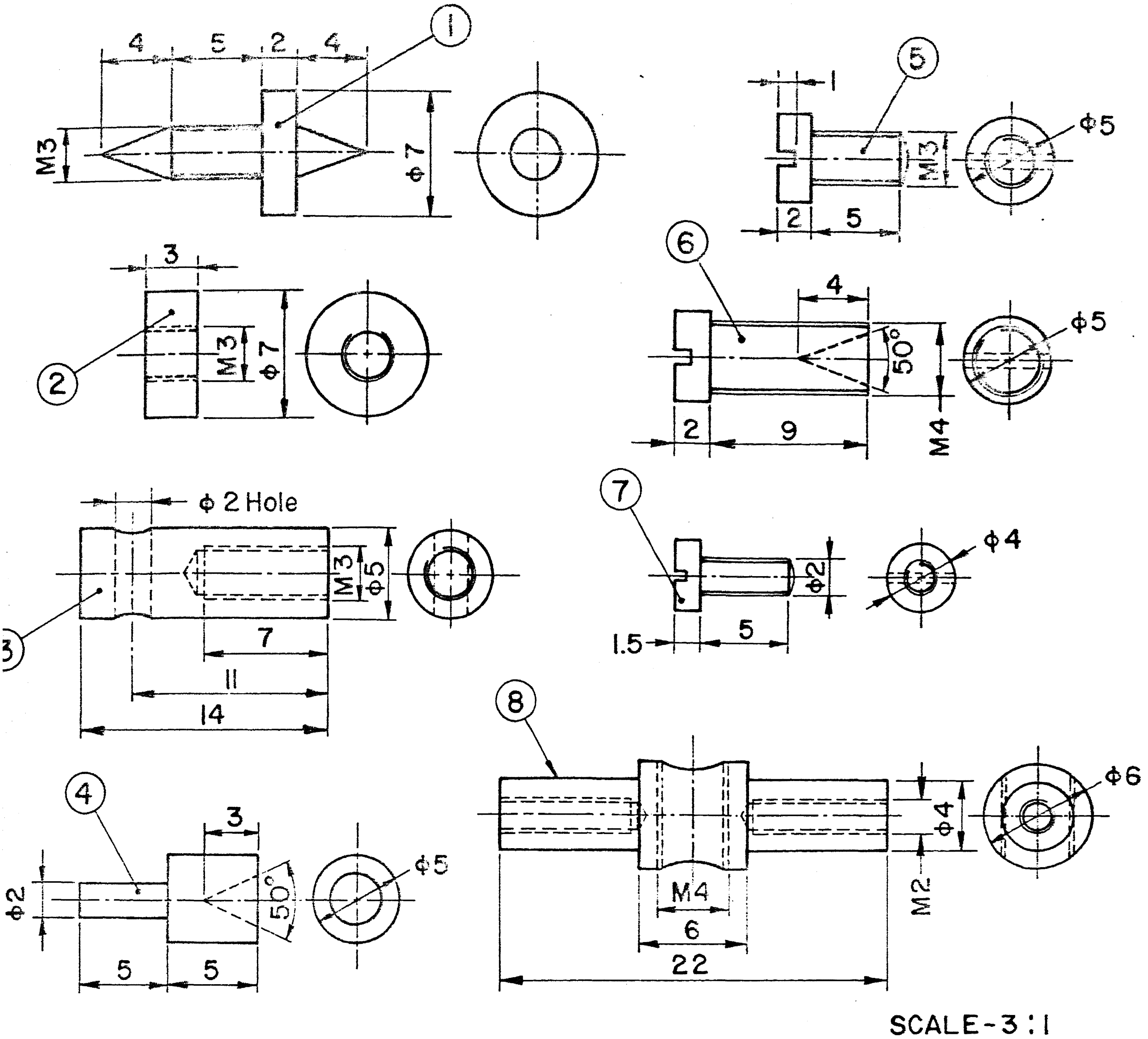
Angle of twist=20 degrees, Clearance between the tube inner wall and outer edge of the blade=2 mm. Propeller blade is shown in Fig. 3.6.

TEMPERATURE MEASUREMENT

Inlet temperature of air is measured with the standard probe of THERM-3280 (Temperature measuring instrument). Temperature of air at various locations inside the test section and the wall temperatures are measured using Copper-Constantan thermocouples. Thermocouples are initially calibrated in a constant temperature bath using the standard temperature probe of THERM-3280.

PRESSURE MEASUREMENT

Small pressure drop across the test section including the entry length is measured by a inclined tube manometer (Angle of



1. Shaft for propeller
2. Nut for part no. 1
3. Supporting leg
4. End support for part no. 1
5. Screw for part no. 3
6. Screw with conical cavity to support part no. 1
7. Screw for part no. 8
8. Supporting leg

Propeller mountings

Figure 3.3

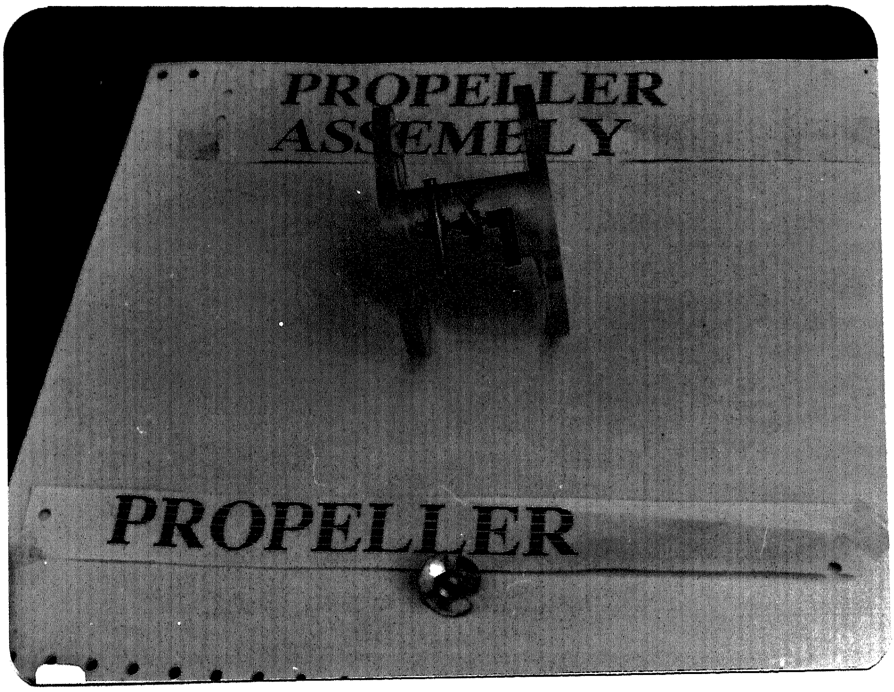


Figure 3.4

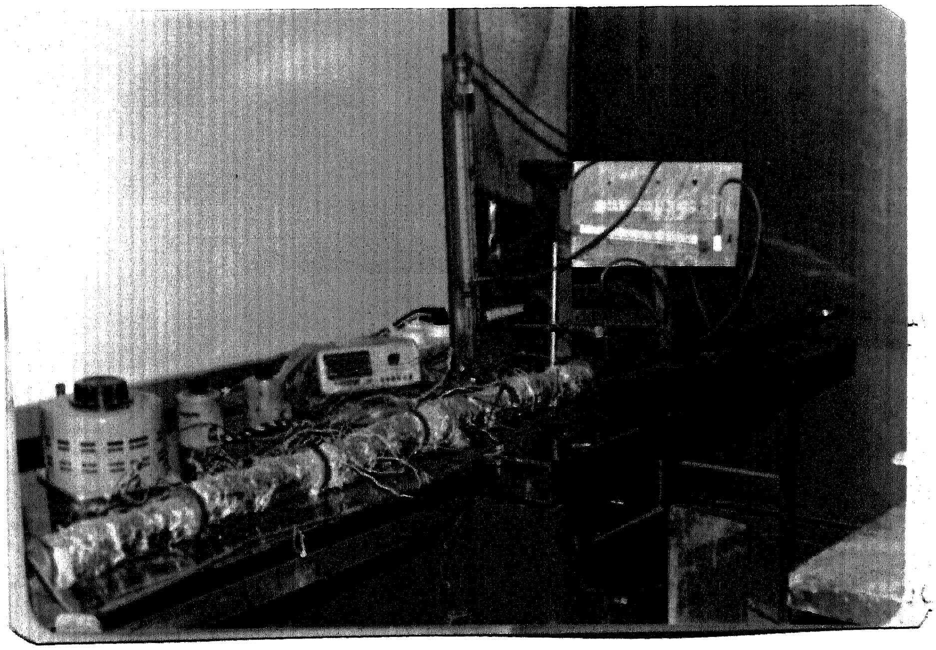
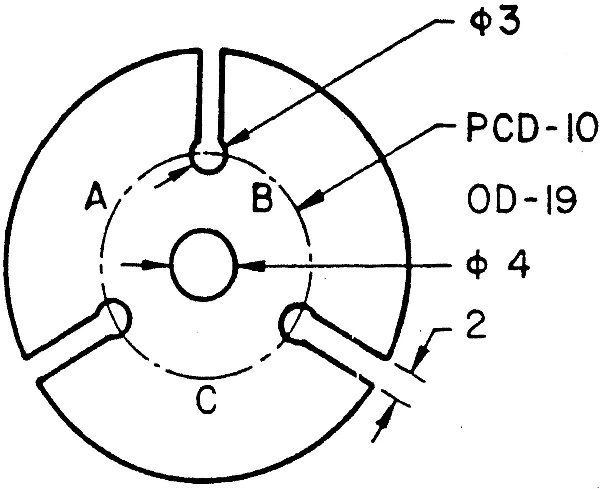


Figure 3.5



A, B, C Blades twisted at an angle of 20°

Fig.3.6 Propeller Blade

inclination=20 Deg.) with dyres oil (Specific gravity=0.826) as a manometric fluid. For higher pressures a U-tube manometer with water as a manometric fluid is used.

FLOW MEASUREMENT

A pitot tube, located at the inlet of the test section, is used to measure the flow velocity. Velocity head is measured using a inclined well type manometer (Angle of inclination=6 Deg.) with water as a manometric fluid.

3.3 EXPERIMENTAL PROCEDURE

Air is used as a test fluid. Steps of the procedure are as follows :-

1. After opening the regulating valve, the blower is switched on and a suitable voltage is applied across the heating tape using a variac.
2. After setting the flow rate at the desired value the pressure drop across the test section including the developing length is measured by a manometer.
3. Velocity head is measured by a manometer connected to the pitot tube.
4. Depending upon the voltage applied across the heating tape and the flow rate, steady state is reached in about two hours and then the various temperatures are measured by thermocouples.
5. All the above steps are repeated for different Reynolds numbers, with and without in-line propellers.

3.4 THE HYDRODYNAMIC DEVELOPING LENGTH

For $Re<2200$, $X_d=0.05.Re.D$, and

For $Re>2200$, $X_d=50.D$, Where X_d=Developing length

Provided developing length=1.4 m

Hence the provided developing length is insufficient for $1000<Re<2200$, in which range no measurements are taken.

CHAPTER 4

RESULTS AND DISCUSSION

4.1 INTRODUCTION

Steady state measurements of wall temperature, inlet and outlet temperature of air, flow rate of air and pressure drop across the test section including the entry length are measured to determine the local heat transfer coefficient and fanning friction factor for flow through pipe subjected to constant heat flux. Experiments are conducted with six in-line propellers, three in-line propellers and no propellers to assess the effect of varying interpropeller distance and Reynolds number on heat transfer augmentation and pressure drop. The experiments are not conducted for low values of Re as the rotation of the propellers does not take place for Re<2300.

4.2 ASSUMPTIONS

1. Considering one dimensional steady state conduction across the test section wall (Brass), the inside wall temperatures are only 0.0024 degree Celsius lower than the outside wall temperatures, the latter being measured by thermocouples fixed at various locations along the length of the test section. This difference is neglected and inside wall temperature is assumed to be equal to the outside wall temperature.
2. Average bulk temperature of the test fluid, T_b is assumed to be equal to the mean of the inlet and outlet air temperatures.

4.3 DETERMINATION OF NUSSELT NUMBER AND FRICTION FACTOR

The local Nusselt number is given by

$$Nu_x = q\ D/(k\ (T_{wx} - T_{ax})) \qquad 4.1$$

Where q, the constant heat flux input is

$$q = Q/A_s$$

$$= \dot{m}\ C_p\ (T_o - T_i)/A_s \qquad 4.2$$

Where,

D = Inside diameter of the pipe,

k = Thermal conductivity of air at T_b

Q = Heat input rate

A_s = Inner surface area of the test section

C_p = Specific heat of air at T_b

$\dot{m}$ = Mass flow rate

Pressure drop across the test section, in terms of the fanning friction factor is given by

$$\Delta p/(\rho_a\ g) = 4\ f\ L_t\ v^2/2\ g\ D$$

$$\text{or} \quad f = \Delta p\ D/(2\ \rho_a\ L_t\ v^2) \qquad 4.3$$

Where,

L_t = Distance between two manometer taps

ρ_a = Density of air

v = Flow velocity

D = Inner diameter of the tube

Reynolds number is given by

$$Re = v\ D/\nu_b \qquad 4.4$$

Here ν_b = Kinematic viscosity of air at T_b

As discussed in Section 2.3, one of the most common methods to

evaluate the performance of a turbulence promoter is by its efficiency defined by

$$\eta = (Nu_e/f_e)/(Nu_s/f_s)$$
$$= (Nu_e/Nu_s)/(f_e/f_s) \qquad 4.5$$

4.4 SAMPLE CALCULATION

DETERMINATION OF Nu_x AND Re

Measured data

$T_i = 23.6\ ^{o}C$

$T_o = 47.5\ ^{o}C$

$T_w = 47.2\ ^{o}C$

$T_a = 27.7\ ^{o}C$

$v = 1.52$ m/s

$\Delta p = 59.8\ N/m^2$

Bulk temperature $T_b = (T_i + T_o)/2 = 35.55\ ^{o}C$

Various property values of air at T_b are

$\nu_b = 1.655$ E-05 m^2/s

$\rho_b = 1.1774\ Kg/m^3$

$k_b = 2.688$ E-02 $W/m\text{-}^{o}C$

$C_p = 1006.0\ J/kg\text{-}^{o}C$

Constants are

$D = 0.0254$ m

$A_c = 5.067$ E-04 m^2

$A_s = 1.21609$ E-02 m^2

Using Eqn. (4.4)

$Re = 2332$

Using Eqn. (4.2)

$q = 150.3 \quad W/m^2$

Using Eqn. (4.1)

$Nu_x = 8.7$

For n = 6, at Re = 10000

$Nu_{fi} = 1.57$

$f_{fi} = 5.73$

$\eta = 0.28$

TABLE 4.1

Measured data for 6 propellers at Reynolds number = 2331

X/D	Wall Temperature	Air Temperature	Nu_x
5	47.2	27.76	8.7
9	51.0	30.5	8.3
19	52.5	33.3	8.8
23	56.9	33.7	8.4
33	57.6	39.9	9.6
37	57.9	42.7	11.1
47	58.1	44.4	12.4
51	59.1	45.5	12.5
61	60.0	46.4	12.4
65	63.3	47.0	10.4

4.5 RESULTS

Fig. 4.1 shows the variation of average Nusselt number, Nu with Re for Number of propellers, n=6, 3, 0 (smooth tube). As the n increases, on the whole, Nu increases. The values obtained for n=6 and 3 are found to be greater than those for a smooth tube. In order to place the propellers in the flow, the test section had to be designed in several sections which were joined by flanges as shown in Fig. 3.2. The data for smooth tube were taken for the

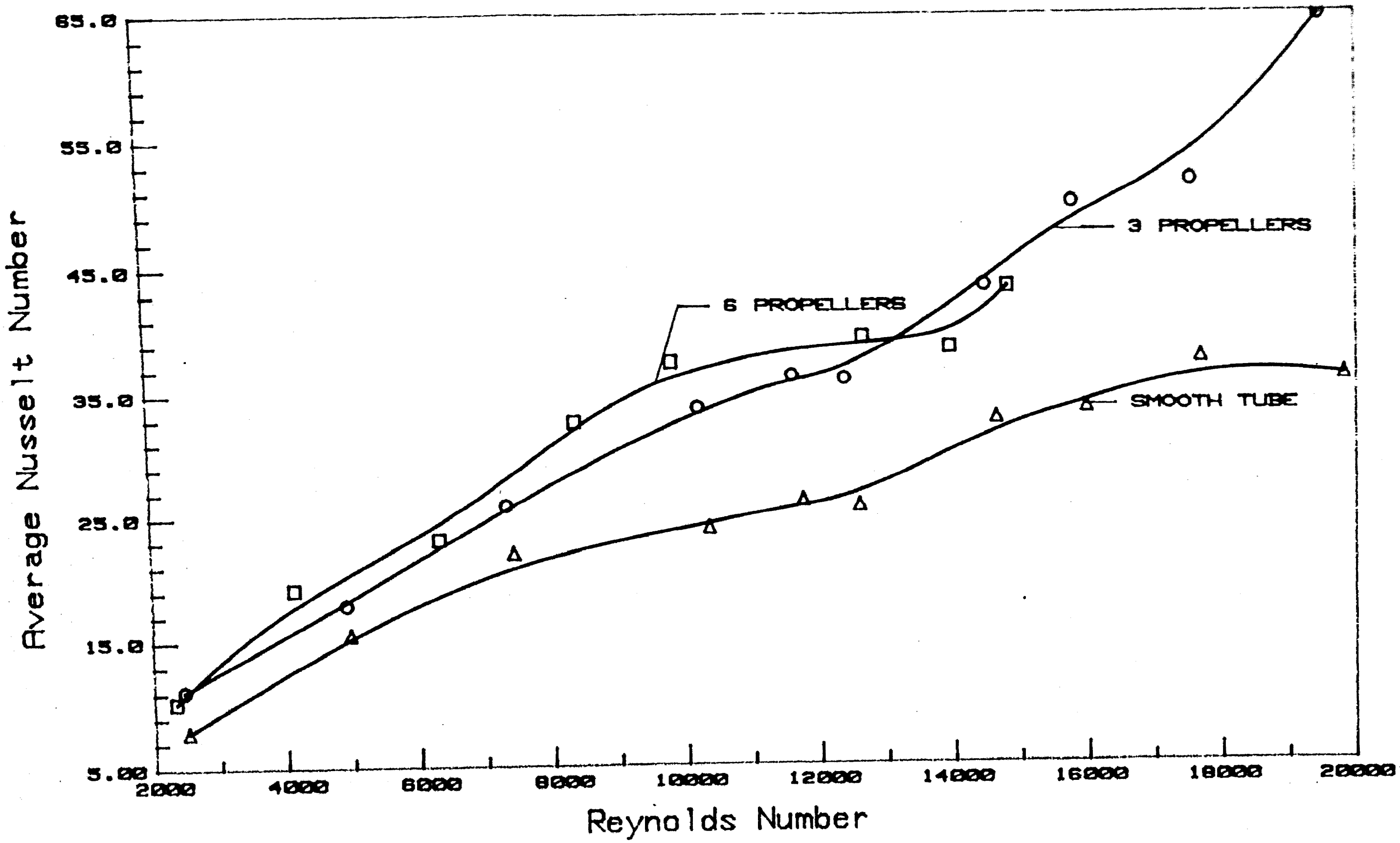


Figure 4.1 Variation of Nu with Re

taken for the test section with propellers removed.

Fig. 4.2 shows the variation with Re of the fractional increase in the average Nusselt number, Nu_{fi}. It is seen from the figure that Nu_{fi} is higher for larger value of n. The curve for n=6 exhibits a maximum around Re=10500 with Nu_{fi}=1.55 and a minimum around Re=6000 with Nu_{fi}=1.3. For n=3, the resistance to the flow is relatively small, resulting in a larger value of the maximum Re=16000 where Nu_{fi} tends to level off with a maximum value of 1.45. The minimum value occurs at Re=6000 which is same as that for n=6 with Nu_{fi}=1.2.

Fig. 4.3 shows the variation with Re of fanning friction factor, f, for n=6, 3 and 0 (smooth tube). As n increases, f increases. As Re increases initially f decreases and then it tends to level off. For n=6, f becomes approximately independent of Re beyond Re=8000. For n=3, f becomes approximately independent of Re beyond Re=12000. For smooth tube, in the beginning f decreases rapidly with Re, then decreases very slowly and levels off around Re=16000. This trend seems to be similar to that for flow through a pipe (White, 1986).

Fig. 4.4 shows the variation with Re of fractional increases in fanning friction factor, f_{fi}. It can be seen from the figure that f_{fi} is larger for larger value of n. As compared to a smooth tube, f_{fi} varies between 2.4 and 7.1 for n=6 and between 2.2 and 3.8 for n=3 for Re<15000. For a large number of enhanced passages of various geometries, f_{fi} lies below 4.5 for Re<15000 (Obot and Esen, 1992). Obviously the pressure drops for n=6 are much higher

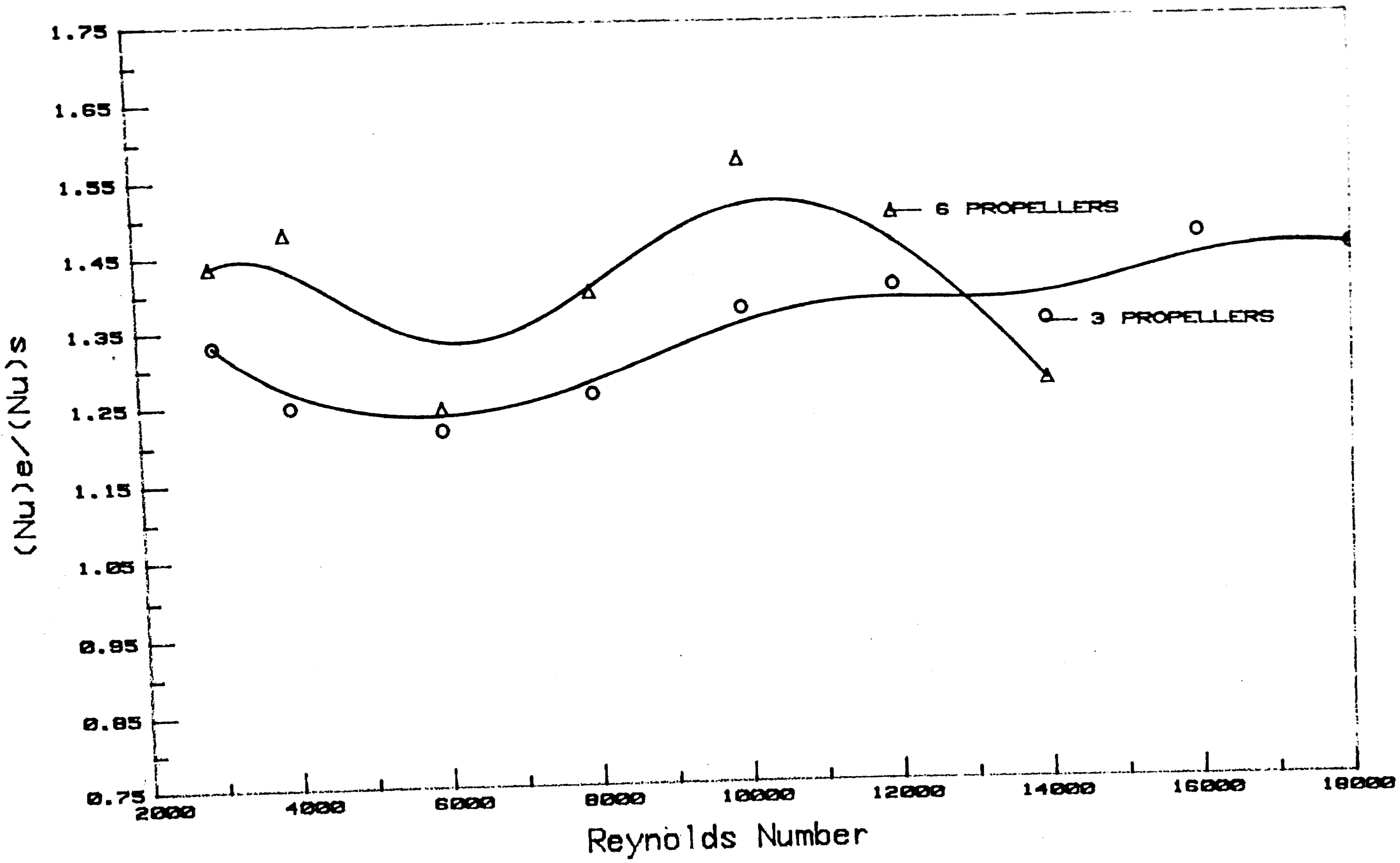


Figure 4.2 Variation of Fractional Increase in Nu with Re

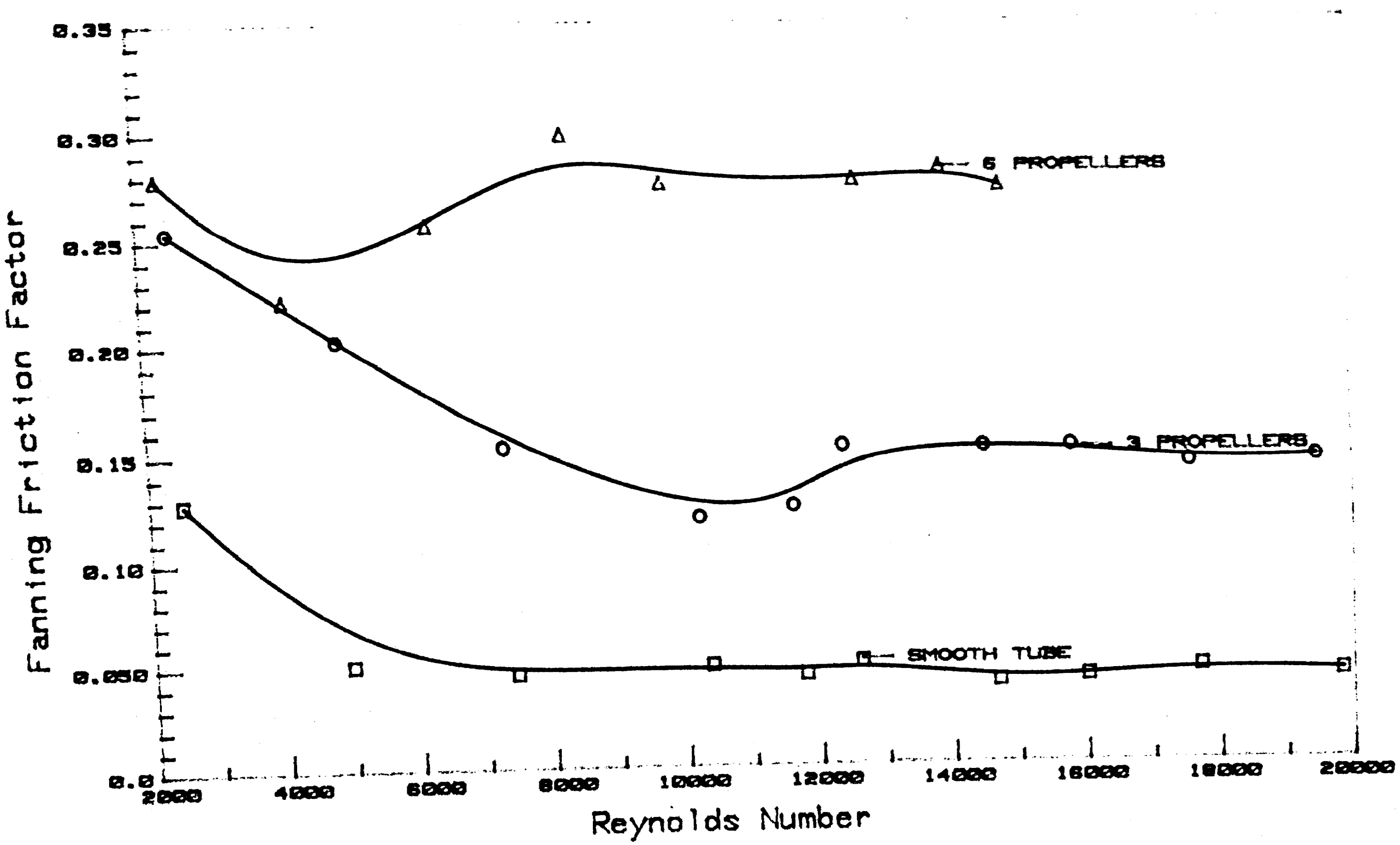


Figure 4.3 Variation of f with Re

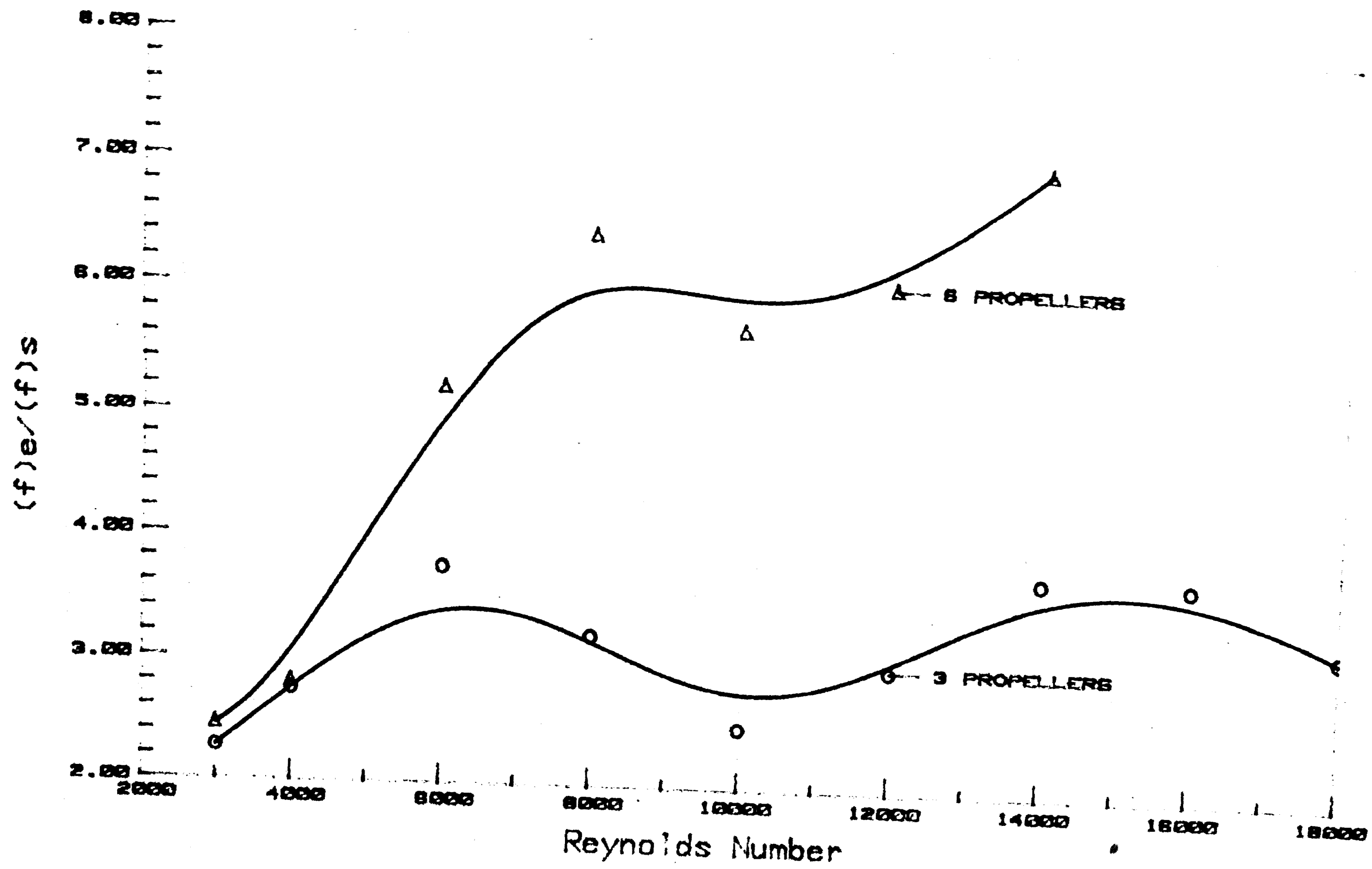


Figure 4.4 Variation of Fractional Increase in f with Re

than those for n=3 and the geometries mentioned in Obot and Esen (1992).

Fig 4.5 shows the variation with Re of the efficiency of the in-line propellers defined by Eqn.(2.1). For n=3 the efficiency is close to 0.58 at Re=3000. It varies with Re showing a maximum of 0.50 around Re=10000 and minimum of 0.35 corresponding to Re=6000 and Re=14000. For n=6, the efficiency decreases with Re and exhibits a value of 0.26 around Re=10000.

A comparison with the augmenting flow passages of Obot and Esen (1992) for Re=14000 (see Table 4.2) shows that only five passages namely GA-3, Y-19, Y-21, Y-22, Y-23 (see Table 2.2) give higher value of Nu_{fi} than that for n=3. The pressure drop with in-line propellers is also much higher than that for any of the flow passages. Due to these reasons the efficiency of the in-line propellers is significantly low compared to any of the aforementioned enhanced flow passages. The data of Chaturvedi and Kant (1992) with water as a test fluid show an efficiency of 2.5 for Re=48000, but it decreases as Re and n decrease. Additional experiments are needed to test the propellers in the whole range of Re with the present design of the apparatus with both air and water to establish their superiority or otherwise. A comparison of the results for n=3 with other passive devices shows that the efficiency value for n=3 is better than that for mesh inserts, brush inserts, disks, streamline shapes, detached promoters and propeller type baffles. The efficiency values for n=3 are closer to the values for wire coils.

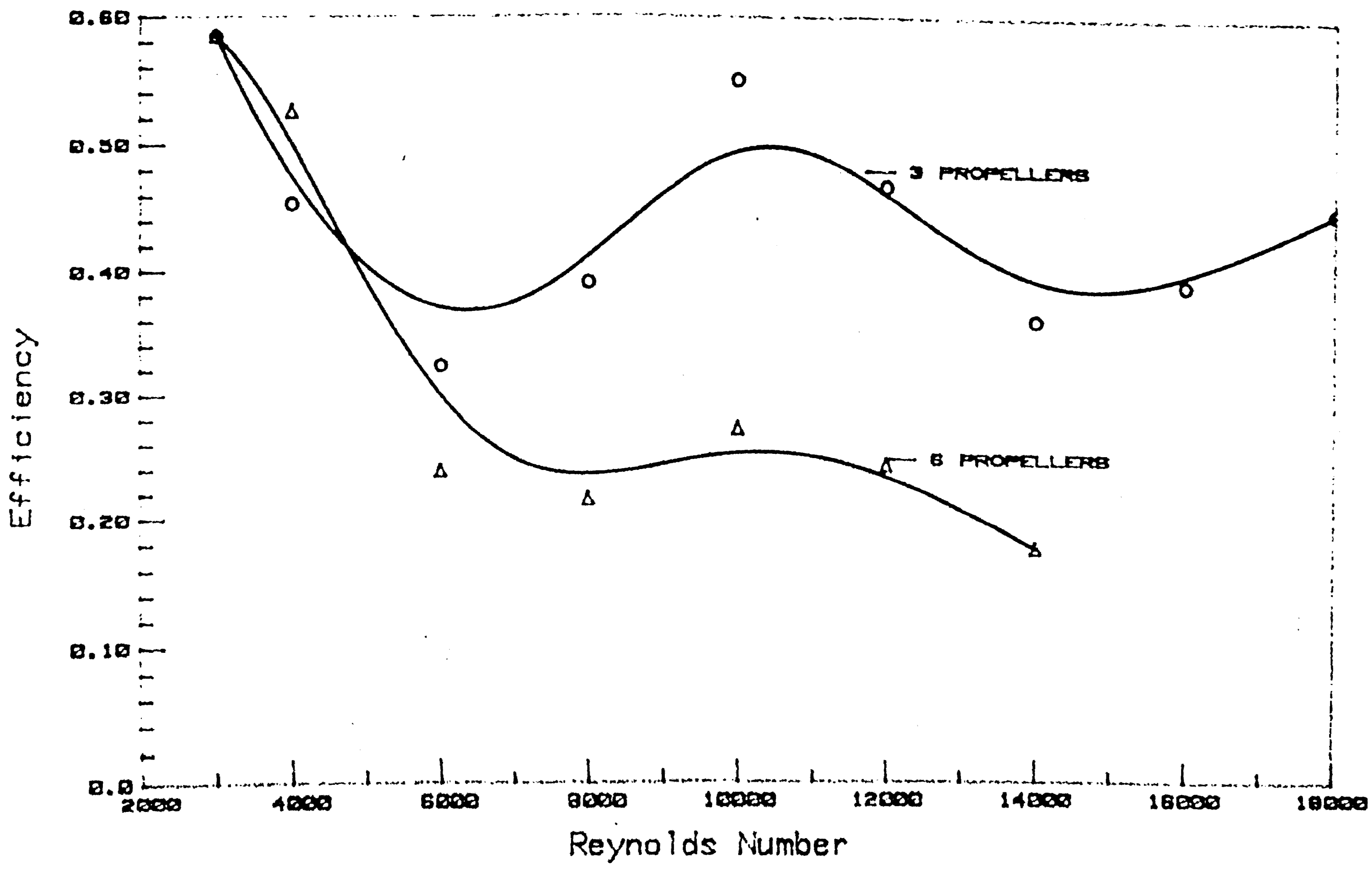


Figure 4.5 Variation of Promoter Efficiency with Re

TABLE 4.2

Comparison of Results at Re = 14000

Flow Passage	f_{fi} *	Nu_{fi} #	η
Obot and Esen			
GA -1	1.6	-	-
GA -2	1.7	-	-
GA -3	1.7	1.8	1.2
HC -4	1.2	1.2	1.0
HC -5	1.7	1.4	0.9
HC -6	1.1	1.3	1.2
W -7	1.4	1.3	0.9
W -8	1.2	1.2	1.0
W -9	1.5	1.3	0.8
W -10	1.8	1.2	0.6
W -11	1.7	1.4	0.9
W -12	1.3	1.0	0.8
W -13	1.7	1.1	0.7
Y -14	1.5	1.1	0.8
Y -15	2.0	1.5	0.8
Y -16	1.4	1.6	1.2
Y -17	1.4	1.6	1.2
Y -18	1.3	1.6	1.2
Y -19	1.9	1.8	0.9
Y -20	1.6	1.1	0.7
Y -21	1.6	1.9	1.2
Y -22	2.1	2.8	1.4
Y -23	2.5	2.1	0.8
Present Work			
n=3	3.6	1.4	0.4
n=6	7.2	1.3	0.22

$Nu_{fi} = Nu_e/Nu_s$

* $f_{fi} = f_e/f_s$

4.6 CONCLUSIONS

In the present study, experimental data are obtained in the range of $2300<Re<20000$.

1. Maximum fractional increase in Nu is found to be 1.55 for n=6 around Re=10500 and 1.45 for n=3 around Re=16000.

2. Maximum fractional increase in f is found to be 7.1 for n=6 around Re=14000 and 3.8 for n=3 around Re=15000.

3. The efficiency value for n=3 is lower than that for twisted tapes and spirally shaped passages (Table 2.2), but it is higher than that for mesh inserts, brush inserts, disks, streamline shapes, detached promoters and propeller type baffles. The efficiency value for n=3 is closer to the value for wire coils.

4. The efficiency for 3 propellers is more than that for 6 propellers.

4.7 SUGGESTIONS FOR FUTURE WORK

1. Experimental study with different test fluids is needed to be carried out to investigate the effect of Prandtl number on heat transfer augmentation.

2. Experiments with in-line propellers should be conducted in a wide range of Re. Due to the limitations of the apparatus the data in the present study were not taken for $Re>20000$.

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